Fundamental And Tribological Analysis of Mechanical Face Seal: A Review

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Abstract:
Earlier centrifugal pump shafts were sealed via rings of elastic packing, squashed by a packing gland, but this sort of shaft seal needs a fair quantity of leakage just to lubricate the packing and keep it cool. Then the use of mechanical face seals began to increase, in which two optically flat surfaces are used out of which one is a primary or rotary ring and the other is a secondary or fixed ring. Predetermined failure of the mechanical seal may occur due to many reasons like improper assembly, mishandling, inadequate lubrication, adhesive and abrasive wear, improper material selection, etc. This paper describes the summary for the construction and working of the mechanical face seal, a detailed discussion regarding the tribological behavior of the seal faces under different lubricating conditions. The significance of hydropads, grooves, and textured seal faces on hydrostatic and hydrodynamic lubrication in sealing gaps has also been discussed along with different heat transfer augmentation techniques.

Keywords: mechanical face seal; tribology; lubrication; wear; friction

1 Introduction

The world pump market is growing at a rate of 3% to 4% per year and centrifugal pumps are sharing 63% of the total pump market [1]. For pumps used in process plants, the sealing system is more reliable and efficient than gland packing [2]. Different types of seals are available nowadays. The mechanical seal can be either single or double type, balanced or unbalanced type, or pusher or non-pusher type. There are mainly three sealing points in a simple mechanical seal, 1. The primary sealing ring (mounted on the shaft through O-ring as shown in fig. 1). 2. Mating ring (sealed against the housing of the pump by using O-ring) and [3]. Both the faces (sealed by making them optically flat within three light bands and pressed against each other by using spring)3. In the arrangement of the seal, any ring from stationary or rotary can be kept flexibly mounted. But in high speed, it is more desirable to keep the rotary ring flexibly mounted as the tracking ability of stationary flexible mounted ring decreases beyond the high speed [4]. The double seal is designed with two primary seals that use two rotating seal faces and two fixed seal faces. Buffer fluid (at a lower pressure than the sealed fluid) or barrier fluid (at a higher pressure than the sealed fluid) is provided between these two seals. This design protects the seal when operating with carcinogenic, explosive, adhesive, or hazardous fluids.

A helical compression spring should be selected such that it provides optimum grip between rotary and stationary seal rings and also compensates the axial movement of the rotary seal ring for axial vibration and carbon face wear [5-6]. This sort of mechanical seal face also needs a few amounts of leakage across the faces to create a hydrodynamic film. This leakage usually evaporates and is not visible. In process plants, most of the process fluids that are sealed by mechanical seal include water, light hydrocarbons, and heavy hydrocarbons. Most process fluids that are sealed by the mechanical seal in industries have a temperature range from

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0 to 50°C and discharge pressure from 1.1 to 5 bar [7-9].

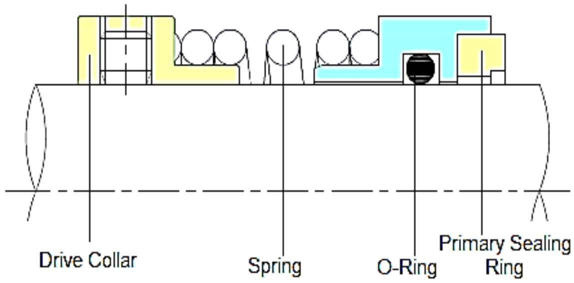


Fig. 1 Mechanical Face Seal

If the sealing gap increases from inside to outside in seal face separation, it is called sealing gap convergence. Sealing gap convergence or coning is generally produced from mechanical and thermal deformation of seal faces. Mechanical face seal can be operated with full fluid film lubrication with sealing gap convergence [10-12]. The inside pressurized seal is operated with negative coning but it cannot support more pressurized loads and results in high contact pressures [13].

Mainly pump shafts today are preserved utilizing mechanical seals. However, because of the fine elements used for this new sealing technique, mechanical seal failures are the biggest reason for pump downtime. There are several reasons behind the seal failure, but when the leakage rate of the pump becomes unacceptable, the seal is considered to have failed [14]. In process plant applications, many mechanical seals fail prematurely [9,15-18]. Ingram [19] has mentioned that up to 70% of incoming pumps for repair are reported due to mechanical seal failure. In 1981 Will [15] stated that his company spends $15 million on operating 14,000 centrifugal pumps per year and 75% of maintenance costs are associated with seal failures. Eeds et. al [3] have described in their article that many different reasons are responsible for the failure of primary seal faces but the main cause out of all of them is loss of lubricant between faces. The study behind fluid film between the seal faces and the tribological behavior of mechanical seal has been always at the center of many researchers.

The lubrication of the mechanical seal becomes worse and the seal wear becomes faster with the increase of surface roughness and decrease of sealed fluid pressure [20]. The flatness error of the seal faces results in the convex shape of the pressure distribution curve in the seal clearance. This leads to an increase in hydraulic force and face-out which ultimately results in a high leakage rate. Inadequate lubrication between the faces can cause failures like adhesive wear, heat checking, and cracking. The unstable lubrication film between the seal faces results in the chipping of the outer edge of the faces. Severe abrasive wear may appear on the seal faces due to abrasives in the pumping fluid as well as due to the crystallization of the product. Often, a pitting effect can be seen on the carbon face due to the selection of the wrong grade of carbon or excess viscosity of the fluid. Failures in the secondary sealing elements are most often found due to improper assembly as well as non-perpendicularity along the shaft of seal faces. Clogging of the spring is possible due to the crystallization as well as the other dirt that has accumulated.

The mechanical face seal consists of a primary ring mounted flexibly on the spring or bellows and so it is almost impossible to have the face of the primary ring completely perpendicular to the shaft. For these reasons, it is important to know the mechanical seal with hydrostatic, hydrodynamic, and thermohydrodynamic behaviors along with diametral tilt and coning and a lot of research has been done in this area during the 1980s to 2000s [21-26]. Along with angular misalignment of any faces of rotation, manufacturing tolerances or seal face waviness and speed above critical speed are some of the factors that cause vibration that has a direct effect on the performance of the seal and the life of the seal [27-28]. At present, the performance and reliability of the seal can be enhanced by using software like FEA and CFD during the design and selection process of a seal [29-30].

2 Material

Generally, soft material (like carbon or graphite) has been taken as a rotary seal face and hard material (like ceramic, silicon carbide, tungsten carbide, stellite, etc.). The deformation of the flexible sealing ring depends greatly on the elasticity modulus of the material. The carbon material is almost universally used as one of the seal faces and can play a crucial role in some fleeting adverse conditions [8]. Mechanical carbon used in seal faces is a mixture of amorphous carbon and graphite [31]. Generally, metal impregnated carbon is used where PV ratings are low (i.e. in unbalanced seals) and resin impregnated carbon is used as one of the face materials where PV ratings are higher and to get better corrosion resistance properties.

Silicon carbide possesses properties like high-temperature strength, excellent chemical resistance, high elastic modulus, excellent abrasion resistance, low thermal expansion, and high thermal conductivity [32]. In misaligned conditions, silicon carbide is a promising element to protect the seal from heat checking. Self-sintered silicon carbide is used where chemical resistance material is required. Ni-WC (Tungsten carbide) is used as a seal material where high strength and high toughness properties are required [31,33-36]. Alumina (Al_2O_3) possesses high hardness, high stiffness, and high strength with excellent dielectric properties. Alumina breaks easily under thermal shock result of rapid heating or dry running. Takami et al [37] had proved based on thermal analysis that mechanical face seal with Hastelloy material performs efficiently in high temperature operating conditions.

Secondary seals are used to seal the primary sealing elements with sleeve or pump housing. Elastomers are rubber-type materials, generally used as secondary seals, can achieve any shape without permanent deformation. Ethylene propylene (EPR), ethylene propylene diene (EPDM), Nitrile (NBR), Fluoroelastomers (FKM), and Perfluoroelastomers (FFKM) are some of the elastomers used as secondary seals considering required properties like aging, ozone resistance, high and low-temperature resistance [31,33]. The deformation of the carbon-graphite ring, temperature amplitudes, and vibration amplitudes have a large effect on the dynamic characteristics of the O-rings/elastomers and hence these all parameters must be carefully considered while designing or selecting secondary seals [38-39].

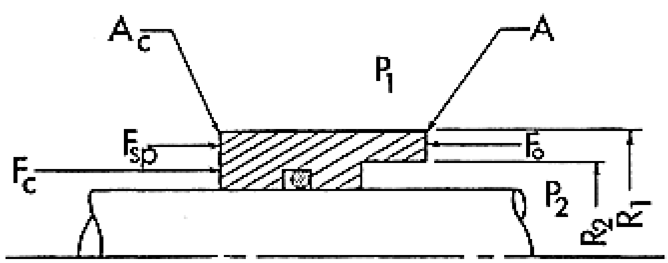
3 Seal Balance & Stability

Fig. 2 illustrates the unbalanced and balanced mechanical seals. As per API 682, the seal balance ratio (B) is calculated as:

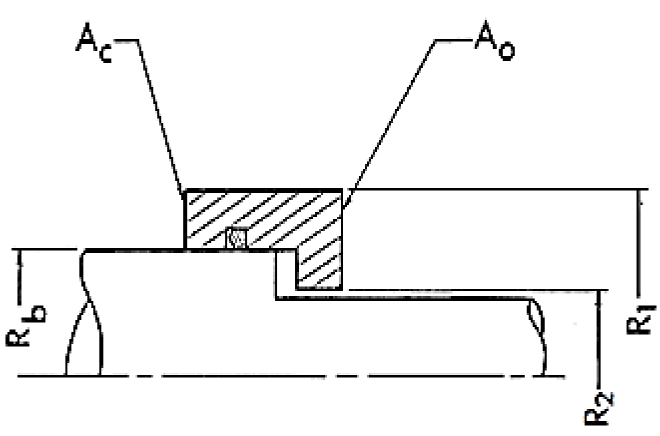
$$B = \frac{\text{Closing area } (A_c)}{\text{Opening area } (A_o)} = \frac{R_1^2 - R_b^2}{R_1^2 - R_2^2} \qquad \ldots(1)$$

A seal is said to be unbalanced when the closing force (hydraulic force caused by stuffing box pressure and spring force) acting on the seal is higher than the opening force (hydraulic force due to fluid/vapor between the seal faces and hydrodynamic lifting force due to non-compressibility of trapped fluid). More heat is generated if the closing forces are the greater forces and unacceptable leakage occurs if the opening forces are the greater forces and that's never desirable. Seal face design and seal balance ratio should be decided such that generated heat and leakage rate at the seal faces can be optimized. Generally, in balanced seals, hydraulic forces acting on the closing area about the face area have been reduced by modifying the shaft design. The unbalanced seal can take pressure from 0.1 to 0.14 kg/mm^2 and the balanced seal can take the pressure up to 0.7 kg/mm^2. Hughes [40] described that seals with narrow faces and low balance ratios are suitable if lesser temperature generation is the primary objective. However, for stability concern high spring loads and high balance ratio is preferred [6, 41].

$$SF = [P_{sp}/\Delta P]/(1 - b) \qquad \ldots(2)$$



(a) Unbalanced Seal



(b) Balanced Seal

Fig. 2 Seal Balance [6]

4 PV Limits

The PV value represents both wear and heat generation and is therefore very important. The factor k is taken as ½ into the equation as the pressure gradient. The value for k may vary from 0 to 1 [41-43].

$$PV = [\Delta P\,(b - k) + P_{sp}]\,V \qquad \ldots(3)$$

The industries are trying to limit the value of PV based on liquid properties as well as seal face material combinations. Lower PV values are given to fluids with lower lubricating properties and higher PV values are given to fluids with better lubricating properties. The PV limit data is often determined by the 3-3/8" size unbalanced seal running in warm water up to 100 hours at 3600 rpm which results in either excessive wear or damage due to thermal overload. The wearing length of the seal is taken as 0.00071” during the 100-hour test (1/8” over 2-year life) to determine the PV limit on wear. However, in the 100-hour test, most of the wear undoubtedly occurs during the first few hours of the test. Some examples of recommended maximum PV values for various material combinations are shown in table 1.

Table 1 Recommended maximum PV values for various material combinations [6]

Material pair	PV value (kpsi.ft/m)
Resin impregnated carbon vs Silicon carbide	5000
Resin impregnated carbon vs Tungsten carbide	1000
Silicon carbide vs Tungsten carbide	600
Silicon carbide vs Silicon carbide	500
Tungsten carbide vs Tungsten carbide	200

5 Tribology of MFS

The thickness of the fluid film generally varies from 0.25µm to 8µm [5]. Heat is generated on the face of the seal by the shearing of the very thin liquid film and sometimes by rubbing the seal faces. This heat is dispersed by conduction through seal rings and from there through boundary layer convection [40]. Flushing is a method used to control the temperature and to prevent the deposition of solids in the area of the mechanical seal. In flushing, relatively cool fluid from an external source is injected in the area of seal sliding faces [44]. Recirculation of the flow of the pumping medium from the discharge piping to the seal is the default flush plan (API Plan 11) for most pumps. Using commercial CFD software such as FLUENT, information about the simulation of the flow field and temperature distribution of seals can be obtained, including heat transfer information in relation to flushing fluid temperature, flushing flow rates, and thermal conductivity of seal rings [45-46]. Keeping the flush rate high does not always give an advantage. Often raising the flush rate also does not increase the cooling. The flow rate of the flush can be determined by simulation of the flow around the mechanical seal ring by CFD analysis [47-48]. As well as the design of the rings and gland can also be improved from this simulation study. Quenching is a method in which pressure-less external fluid is feed on the atmospheric side of the seal faces [5]. Quenching helps to monitor the leakage rate of the seal as well as prevents the pump from dry running.

5.1 Lubricating Film

Surface roughness is responsible for the generation of a lubricating film between the mechanical seal faces. This thin film needs to be retained throughout life to meet the long operating service life of the mechanical seal. The mean waviness of the metal seal face (silicon carbide or tungsten carbide) is generally kept between 0.25-0.45µm and the waviness of the carbon-graphite is kept between 0.05-0.2µm for the development of a good lubricating film. Higher the value of surface roughness on metal seal face results in excessive wear of carbon seal face [34,49]. However, due to the influence of surface roughness, with an increase in velocity, seal faces can also be completely separated from the fluid film due to the hydrodynamic effect. For seal face lubrication, the viscosity of the liquid inside the seal face cavity should be sufficient so that the face could not damage during start-up. Low viscosity fluids are poor lubricants and higher viscosity fluids may cause sticking of the seal faces during start-up. Ideal fluid for seal lubricant should possess a viscosity greater than 14 cSt and less than 32 cSt [50].

The thickness of the lubricating film developed between the seal faces lies in the fraction of micron and hence under certain operating conditions face-seals do not operate with complete separation but in mixed lubrication. The film between the rotating and stationary seal faces can break during the working condition and there is solid contact between the seal faces and the resulting heat and thermomechanical effects can lead to cracking on the sealing surface [34].

Hydrodynamic pressure is generated in the fluid film due to the effects of waviness, misalignment, and vibration of the faces [51-53]. Much research work has been done by Lebeck [54-56] on seal behavior under mixed friction using hydrostatic and hydrodynamic seal face models. The coefficient of friction is a function of mechanical pressure, the nature of fluid contact between the faces, and some other factors, and hence it is not constant. Doust and Parmar [57] had developed a model to study the effects of face convergence, spring pressure, and secondary seal friction on the hydrostatic fluid film thickness. In which it was found that secondary seal friction produces a hysteresis effect in the predicted relationship of the film thickness versus pressure. Zeus [58] has described that the total power consumption of a seal is the result of various power losses in the different sections (churning between concentric cylinders, churning in the disk area, and viscous shear of the interface film) of the seal. Due to the modification of the running condition of the fluid flow in the sealing dam, transition from laminar to turbulent results in an increase of dissipated power but a lower leakage rate. Non-laminar flow has a beneficial influence on damping and hydrodynamic stiffness [59]. Brunetière et al [60-61] had developed a numerical three-dimensional model of a face seal. Later they used this numerical model and did a parametric study of the noncontact liquid face seal. It was found that with increasing the rotation velocity and fluid viscosity of the seal, the temperature of the film increases, and consequently the taper of the faces and film thickness increases which is favorable for good lubrication of the seal interface. At the same time, the leakage rate also increases but it is negligible. Dynamic whirl arises from the hydraulic tilt instability when the balance ratio is less than the critical value and is controlled hydrodynamically as well as by elastomer. It was observed by Metcalfe [62] that suppression of the whirl produced in hydrodynamic seal depends on the stiffness and damping capacity of the elastomer.

Different methods have been published in various articles for measuring the thickness of liquid film that includes electric resistance method [63-64], voltage drop method [65], capacitance method [51,66], eddy current proximity method [67], and ultrasonic reflection method [68-70]. Acoustic emission technique had been adopted by Zhang et al [71] and then the recognition model was built via cascaded decision by which the film thickness can be estimated. In 2018, Towsyfyan et al [72] had developed a mathematical model which can predict the energy of an AE signal under different tribological regimes which provides a reliable condition monitoring system.

5.2 Change of Phase

If the fluid passing through the seal is close to the saturation state, the change in phase will occur from liquid to vapor as a result of the immediate temperature rise and pressure drop [73]. Buck [74-75] showed that seal reliability is highly dependent on the proportions of liquid and vapor between the seal faces and also suggested a method for estimating amounts of liquid and vapor between seal faces.

Cavitation and vapor transitions both represent the same situation in which the film pressure goes below the critical pressure. For vapor transitions, the pressure at local temperature is always a vaporization pressure, while the pressure for cavitation can be either an air saturation pressure of the film or vaporization pressure (boiling point) that depends on the local temperature. In the case of vapor transitions, it can be said that the cavity moves to the edge and allows the vapor phase to exit the interface. Vapor flows continuously through the interface so that the cavity is in the dynamic equilibrium state while in the cavitation, the cavity is inside the fluid film [76].

Liquids like light hydrocarbons easily vaporize at atmospheric pressure and temperature and thus maintaining a lubricating film throughout the life of the seal is difficult. Hughes et al [54] had shown phase-change effects in parallel and tapered liquid face seals analytically through both isothermal and adiabatic models. Working fluid temperature also affects the phase change in the mechanical face seals. In 2015 Brunetière et al [77] had developed a transient numerical model using the homogeneous fluid theory to analyze the effect of change in feeding water temperature on vaporization of the fluid film in a mechanical seal interface. It was concluded that if there is a moderate increase in the temperature rise, there is no significant effect in seal performance due to vaporization of fluid, but unstable behavior was observed in the mechanical seal over the high temperature.

5.3 Thermal Behavior

In mechanical face seals, heat is generated either due to the shearing of the fluid film or due to the rubbing contact between the sliding faces. For heat flow, there must be a temperature difference between the seal faces, the liquid film as well as the product temperature around the seal rings. It is possible to measure the temperature at which the liquid film vaporizes under a given seal operating conditions (speed and pressure) experimentally. Temperature distribution on the seal faces can be predicted by the Nusselt number. Many numerical analyses have been proposed for the estimation of the heat transfer of the mechanical face seal and also the experimental work has been performed using an infrared camera to validate it [78-81].

Thermal deformation is more significant than mechanical deformation. Thermal deformation distorts original flat sealing surfaces along a diameter to a convex shape, that can have amplitude up to 2 µm for seal ring and 0.75 µm for seat [82]. This thermal deformation promotes ID wear. Due to this non-uniform wear, the seal face loses flatness, and line contact can be formed between the seal faces instead of the surface contact, and hence the leakage rate might get increased. Whenever thermoelastic instability occurs in a seal, patches are found on the distressed surface of the seal in which hotspot detects near center and results in a higher wear rate of the seal face [83-85]. When the seal is operated with a light-specific gravity liquid, the thermoelastic instability of the seal face is quite common, which can also be experienced with audible sound near the stuffing box area. During this time primary ring goes under additional axial motion parallel to the shaft axis and it results in the wear of other parts such as the seal's anti-rotation device. In studies concerning thermal distortions of seal elements, the seal interface is considered to be isothermal as seal rings are generally made of conductive materials, and this assumption is used in Reynolds equation solutions [86]. Hughes et al [73] had proposed an adiabatic model by considering seal plates as insulators. To understand thermoelastic deformation it is necessary to have an idea about temperature generated due to intermittent contact between the seal faces in the sealing system and it can be measured using thermocouple [78,87-88]. In 2013, Gupta et al [89] had suggested a permanent magnet sensor for the measurement of seal face temperature. Above mentioned techniques are very easy to use but they give measurements only at local points and their response is not well adapted to dynamic measurements. Infrared thermography is a technique enabling to the determination of temperature without contact, it is adaptable to dynamic measurements and permits thermal mapping [90].

Brunetière et al [91-92] introduced several models one by one to understand the thermoelastohydrodynamic (TEHD) behavior of mechanical seals near the year 2010. It has been shown that the mechanical seal is controlled by six non-dimensional parameters, four of which arise from contact of asperities (elastic parameter, dimensionless asperity height, dry friction temperature, and reduced balance ratio). The last two parameters are the sealing number and the coning number, which are sufficient for describing its behavior when there is no contact between the seal faces. After that, a multiscale approach for mixed lubrication was introduced as the deterministic approach takes more time in computation, in which the information of roughness induced pressure generation was provided. Subsequently, the seal's behavior for differential lubrication regime was held experimentally and friction torque, temperature, and leakage were measured and it was found to have a strong agreement with the results of the numerical model [93-96].

6 New Developments in MFS Design

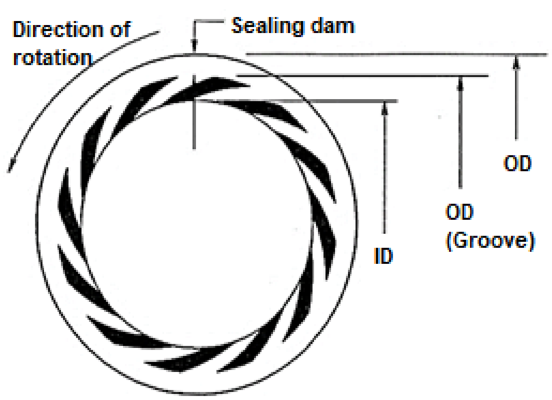
In mechanical seals, the total load is supported by hydrostatic, hydrodynamic or partial contact of the seal face. Due to the intermittent contact between the seal faces, the seal reliability of the seal depends on whether the proper fluid film between the seal faces is retained or collapsed. The lubricating film between sealing surfaces is required to reduce and maintain wear. At the same time clearance between the faces must be kept very small (say, less than10-4 inches) to minimize leakage [97]. It was seen that leakage rate and torque increase due to thermal distortion of seal faces due to heat generation or variations in surface flatness during manufacturing [98]. Asperity friction is the major source of heating in gas seals while viscous shearing is a major source of heating in liquid seals [99]. The wear properties of the seal can be increased by applying coating on the faces. Diamond coating provides good sealing durability due to reduced friction and wears rate on the seal face but it is used rarely as the diamond coating is very expensive and it is very difficult to mold in complex geometry [100-102].

The increased misalignment between the seal face is responsible for the intermittent contact and the seal failure caused by friction. To improve the seal life, the contact elimination method for reducing misalignment by controlling the air pressure according to the clearance between the faces which can be measured by eddy current proximity probes had been discussed in some articles [103-108]. Monitoring the content of this fluid film is always beneficial to take preventive actions. Salant et al [109-110] had introduced a method for measuring contact between the seal faces by reflexed ultrasonic waves produced from a piezoelectric transducer attached behind the stationary seal face. Golikar and Hirani [111] had prepared a test setup for online wear monitoring of mechanical face seal using torque sensor through which the failure of seals under various rotational speeds, lubrication mechanisms, and spring loads can be predicted.

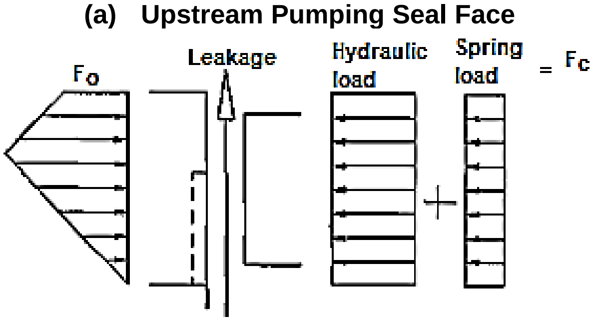
Results show that in the flat face seal, the state of operation is changed suddenly but this does not happen in the wavy seal. With the radial taper, if the waviness is provided on the seal face, then the friction and wear on the face of the seal can be greatly reduced. It also provides sufficient hydrodynamic (due to cyclic variations) load capacity due to squeezing film action so that seal life is also increased [112-114]. Young et al [115] stated in 2003 that providing a wave pattern on a sealing face of the mechanical seal from a laser micro-machined increases the reliability of the seal and reduces the leakage rate. Even if we generate a wavy profile like radial taper, the lubricating film thickness is very low, and the wavy profile could destroy quickly due to some factors like pressure pulsation, vibration, dry running, and solids in the liquids. To solve the seal face problems like high heat generation, poor lubrication, abrasion, and barrier system complexities, many new face designs had been developed that include seal face with hydropads, upstream pumping seal, seal face with texture or micropores, seal face with oriented dimples, etc.

6.1 Hydropads and Grooves

Hydropads generally work to improve load support through the hydrodynamic generation of elevated pressure within the lubricating film. After maintaining adequate film thickness, when the sliding contact between the primary and mating rings of the seal begins then the surface waviness of both the faces increase due to frictional heat. The increase in this surface waviness is due to the small thermal deformation. Netzel [116] showed that seal life can be increased by reducing the frictional heat that occurs if a hydropad or lubrication recess is provided in a mechanical seal. Hydropads can avoid intense localized heating that results in surface disturbance and reduces seal life [117].



(a) Upstream Pumping Seal Face



(b) Opening and Closing Force

Fig. 3 Upstream Pumping [119]

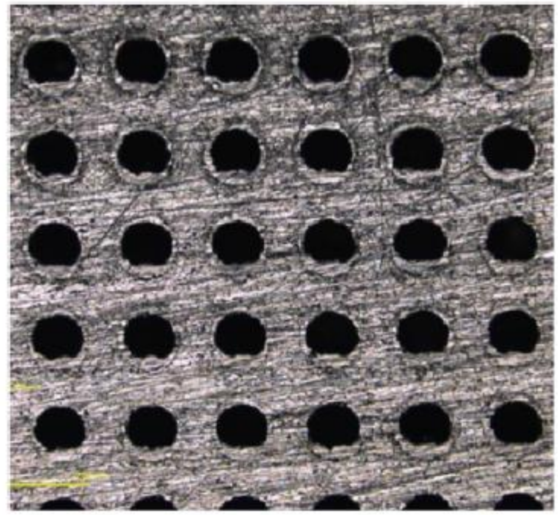
Upstream pumping seal was first discussed by john crane engineers in which spiral grooves are provided on the rotary face of the seal as shown in fig. 3(a). The barrier fluid inserts into the spiral grooves and with rotation of the seal face, fluid moves outward toward the closed end of the grooves due to the viscous shear. It is believed that the spiral groove does not allow the entered fluid to exit from the groove tip and hence pressure is generated and ultimately the opening force is increased [118]. In a pure hydrostatic seal, the generated sealing gap depends only on pressure and does not depend on the speed, but in upstream pumping, generated sealing gap depends on both pressure and speed, and therefore it is both hydrostatic and hydrodynamic [119-120]. When the spiral groove seal is used, the opening force is the sum of the pressure drop across the face and the pressure generated due to the spiral groove pattern and the closing force is the sum of system pressure acting behind the face of the seal and the spring force as shown in the fig. 3(b).

The seal face with the spiral grooves acts as a pressure generating system upon rotation. It can be said that the oblique or spiral grooves in the radial direction will get sinusoidal variation in the circumferential direction of the film thickness, which serves as a wavy seal face. Spiral groove patterns are used to increase load support and control leakage in gas seals. A shallow oblique or spiral groove pattern on the face of a seal can reduce leakage rate if properly designed. Speed, radius ratios, coning, groove depth, groove Inclination, and the number of grooves are important design parameters [121]. Qiu & Khonsari [122] performed a three-dimensional thermohydrodynamic analysis of the spiral groove mechanical seal and discovered that as the groove depth increases from 20µm up to 60µm, the seal temperature as well as the leakage rate decreases. The problem of fouling may occur in shallow grooves or hydropads due to contaminants and hence deep hydropads and grooves can also be used frequently [117].

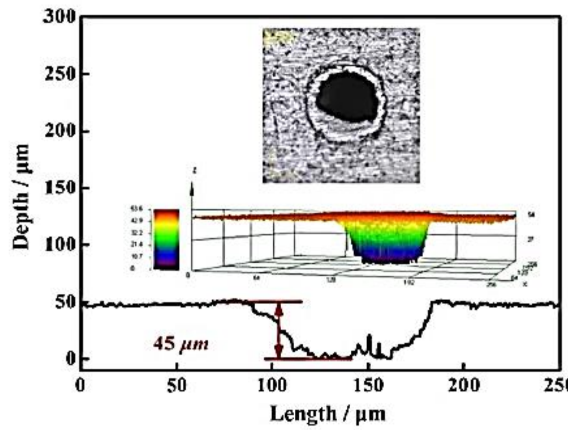
6.2Surface Texturing

After hydropads and grooves on the mechanical seal face, research was conducted on seal opening pressure, fluid film stiffness as well as friction performance, by providing micropores through laser texturing and it was found that the seal with laser texture offers lower temperature rise, lower frictional torque, and improved fluid film stability [123-125]. Microgeometris can also be provided on the surface of the mechanical face seal with the etching technique. These micro geometries can be made of different patterns and depressions with different depths. In 1999, Etsion et al [123] found that the effect of pores depth over diameter ratio is more significant than the effects of area density of the pores and radius ratio of the seal. To obtain the improved seal performance through textured seal compared to conventional un-textured seal, hydrodynamic effects due to speed, viscosity, and seal clearance should be much greater than the hydrostatic effects. Apart from lubrication entrapment and debris trap, the surface texture is also used for lubricant flow control. In 2012, Brunetière et al [126] compared the performance of rough-textured surfaces with smooth textured surfaces. The authors tested the effect of texture density and aspect ratio as well and showed that rough-textured surfaces give a significant reduction in friction. Zhang et al [127] obtained friction coefficient and wear rate for the micro surface textured sealing ring using pin-disk tribology tester and found that micro surface textured sealing ring offer good tribological properties. Fig. 4 depicts the micro surface textured sealing ring. Shen and Khonsari [128] investigated experimentally and numerically the effect of dimple internal structural shapes rectangle, oblique triangle and isosceles triangle on hydrodynamic lubrication. They found that the cylindrical dimple always generates more load-carrying capacity than dimples with triangular profiles under different working conditions.

Meng et al [129] tested the feasibility of leakage control by providing evenly distributed oriented dimples of different shapes (diamond, ellipse, rectangle, and triangle) on mechanical face seals. It was concluded that the rectangle-oriented dimples have the largest reverse pumping capability. Adjemout et al [130] analyzed the real dimple shapes produced using low-temperature plasma coupled with a thermos chemical surface treatment on stainless steel sealing rings. It was mentioned that defects like roughness in the dimples can dispel the textures' positive effects. Hence, it is very hard but important to control the shape and geometry of the dimple. But in 2019 Wang et al [131] proved that optically free form textures provide a better coefficient of friction and leakage performance than optimal circular dimples.



(a) Micro Surface Textured Dimple



(b) Parameters of Dimple

Fig. 4 Micro Surface Textured Ring [127]

6.3Heat Transfer Augmentation

Excessive heat generated at the seal face interface is the main cause of mechanical seal failure. In the last few decades, various attempts have been made to remove heat from the seal interface to reduce the interfacial temperature and increase the life of the mechanical seal. Flushing is a method used to control the temperature of solids in the area of the mechanical seal. Xiao et al [132] had reviewed the different patents on mechanical face seals that can be used for heat transfer augmentation.

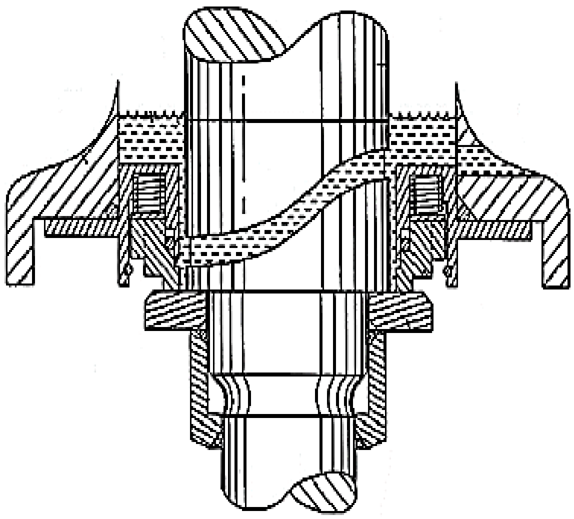


Fig. 5 A Vertical Pump Gear Box Seal with Oil [134]

In 1989, Drumm [133] designed a mechanical seal with a heat exchanger in which both rings are completely surrounded by the annular heat exchanger. There is no flush plan and there is a thin annular fluid zone between sealing rings and heat exchanger in the design. The generated heat at the interface is dissipated through the thin fluid zone into the heat exchanger. In this design the convective heat transfer near the interface is low. Young [134] patented a mechanical seal assembly in which a continuous groove is milled into the shaft sleeve as shown in fig. 5 claiming that the design can circulate the fluid more effectively through the housing and that leads to reduced face temperature.

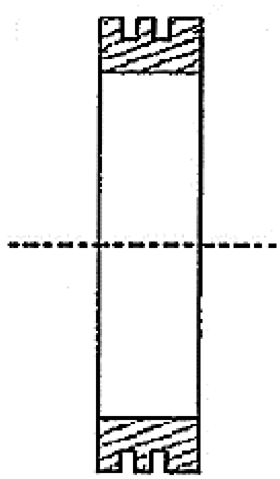


Fig. 6 A stationary ring with circumferential grooves [135]

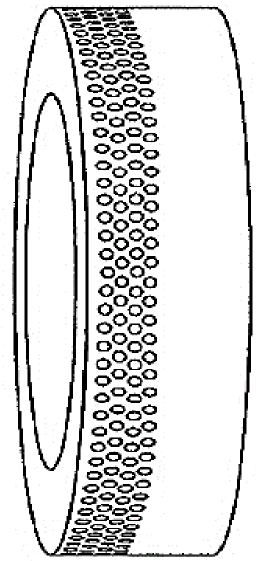


Fig. 7 Dimpled surface stationary ring [139]

In 2008, Khonsari and Gidden [135] claimed that providing a circumferential groove as shown in fig. 6 can enhance the heat transfer and reduce the interface temperature. Dimples provided on the sidewall of the outer diameter of the seal as shown in fig. 7 offers reduced seal interface temperature, and it was proved that providing dimples on the sidewall of the mating ring increases the solid-fluid contact area and gives an effective heat dissipation. Providing dimples can reduce the interface temperature by up to 10%. Dimple depth over dimple diameter ratio is a very important parameter for enhancing seal performance [136-139].

7 Conclusion

The mechanical seal's premature failure has always been the main reason behind the keen area of interest of many researchers. Lots of research work has been done on different face material combinations, development in seal face design, and heat transfer augmentation techniques. Lesser heat should be generated at the seal interface and faster heat transfer of that generated heat are the important factors in the formation of any mechanical face seal design. This article briefly describes the fundamentals and working of mechanical face seals as well as the various changes suggested by different authors in seal design to improve the tribological behavior and to increase the heat transfer capacity of the seal. Ideally, the design of a good mechanical face seal should be such that it can be installed easily and from which dramatically improved seal service life can be obtained. Any further development in the design of mechanical face seal in which seal stability and seal opening force can be increased and hence ultimately the tribological behavior of the seal can be enhanced will be always beneficial for the seal industries.

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